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(54) Title: ACOUSTIC DEVICE

(57) Abstract

    From one aspect the invention is an acoustic device, e.g. a loudspeaker, comprising a resonant multi–mode acoustic radiator panel having opposed faces, a vibration exciter arranged to apply bending wave vibration to the resonant panel to produce an acoustic output, means defining a cavity enclosing at least a portion of one panel face and arranged to contain acoustic radiation from the said portion of the panel face, wherein the cavity is such as to modify the modal behaviour of the panel. From another aspect the invention is a method of modifying the modal behaviour of a resonant panel acoustic device, comprising bringing the resonant panel into close proximity with a boundary surface to define a resonant cavity therebetween.

5   TITLE: ACOUSTIC DEVICE

10   DESCRIPTION

15   TECHNICAL FIELD

The invention relates to acoustic devices and more
particularly, but not exclusively, to loudspeakers
incorporating resonant multi-mode panel acoustic radiators,
e.g. of the kind described in our International application
20 WO97/09842. Loudspeakers as described in WO97/09842 have
become known as distributed mode (DM) loudspeakers.

Distributed mode loudspeakers (DML) are generally
associated with thin, light and flat panels that radiate
acoustic energy equally from both sides and in a complex
25 diffuse fashion. While this is a useful attribute of a DML
there are various real-world situations in which by virtue
of the applications and their boundary requirements a
monopolar form of a DML would be preferred.

In such applications the product may with advantage be light, thin and unobtrusive.

BACKGROUND ART

5 It is known from International patent application WO97/09842 to mount a multi-mode resonant acoustic radiator in a relatively shallow sealed box whereby acoustic radiation from one face of the radiator is contained. In this connection it should be noted that the term 'shallow' in this context is relative to the 10 typical proportions of a pistonic cone type loudspeaker drive unit in a volume efficient enclosure. A typical volume to pistonic diaphragm area ratio may be 80:1, expressed in ml to cm^2. A shallow enclosure for a resonant panel loudspeaker where pistonic drive of a lumped air 15 volume is of little relevance, may have a ratio of 20:1.

DISCLOSURE OF INVENTION

According to the invention an acoustic device comprises a resonant multi-mode acoustic resonator or radiator panel having opposed faces, means defining a 20 cavity enclosing at least a portion of one panel face and arranged to contain acoustic radiation from the said portion of the panel face, wherein the cavity is such as to modify the modal behaviour of the panel. The cavity may be sealed. A vibration exciter may be arranged to apply 25 bending wave vibration to the resonant panel to produce an acoustic output, so that the device functions as a loudspeaker.

The cavity size may be such as to modify the modal

behaviour of the panel.

The cavity may be shallow. The cavity may be sufficiently shallow that the distance between the internal cavity face adjacent to the said one panel face and the one

5 panel face is sufficiently small as to cause fluid coupling to the panel. The resonant modes in the cavity can comprise cross modes parallel to the panel, i.e. which modulate along the panel, and perpendicular modes at right angles to the panel. Preferably the cavity is sufficiently

10 shallow that the cross modes (X,Y) are more significant in modifying the modal behaviour of the panel than the perpendicular modes (Z). In embodiments, the frequencies of the perpendicular modes can lie outside the frequency range of interest.

15 The ratio of the cavity volume to panel area (ml:cm^2) may be less than 10:1, say in the range about 10:1 to 0.2:1.

The panel may be terminated at its edges by a generally conventional resilient surround. The surround may resemble the roll surround of a conventional pistonic

20 drive unit and may comprise one or more corrugations. The resilient surround may comprise foam rubber strips.

Alternatively the edges of the panel may be clamped in the enclosure, e.g. as described in our co-pending PCT patent application PCT/GB99/00848 dated 30 March 1999.

25 Such an enclosure may be considered as a shallow tray containing a fluid whose surface may be considered to have wave-like behaviour and whose specific properties depend on both the fluid (air) and the dimensional or volume box

geometry.   The panel is placed in coupled contact with this active

wave surface and the surface wave excitation of the panel excites the

fluid.   Conversely the natural wave properties of the fluid interact

with the panel, so modifying its behaviour.   This is a complex coupled

5   system with new acoustic properties in the field.

Subtle variations in the modal behaviour of the panel may be

achieved by providing baffling, e.g. a simple baffle, in the enclosure

and/or by providing frequency selective absorption in the enclosure.

From another aspect the invention is a method of modifying the

10   modal behaviour of a resonant panel loudspeaker or resonator,

comprising bringing the resonant panel into close proximity with a

boundary surface to define a resonant cavity therebetween.

BRIEF DESCRIPTION OF DRAWINGS

Figure 1 is a cross section of a first embodiment of sealed box

15   resonant panel loudspeaker;

Figure 2 is a cross-sectional detail, to an enlarges scale, of

the embodiment of Figure 1;

Figure 3 is a cross section of a second embodiment of sealed box

resonant panel loudspeaker;

20   Figure 4 shows the polar response of a DML free-radiating on

both sides;

Figure 5 shows a comparison between the sound pressure level in

Free Space (solid line) and with the DML arranged 35mm from the wall

(dotted line);

25   Figure 6 shows a comparison between the acoustic power of a DML

in free space (dotted line) and with a baffle around the panel between

the front and rear;

Figure 7 shows a loudspeaker according to the invention;

Figure 8 shows a DML panel system;

Figure 9 illustrates the coupling of components;

Figure 10 illustrates a single plate eigen-function;

Figure 11 shows the magnitudes of the frequency
response of the first ten in-vacuum panel modes;

Figure 12 shows the magnitudes of the frequency
response of the same modes in a loudspeaker according to
the embodiment of the invention;

Figure 13 shows the effect of the enclosure on the
panel velocity spectrum;

Figure 14 illustrates two mode shapes;

Figure 15 shows the frequency response of the
reactance;

Figure 16 illustrates panel velocity measurement;

Figure 17 illustrates the microphone set up for the
measurements;

Figure 18 shows the mechanical impedance for various
panels;

Figure 19 shows the power response of various panels;

Figure 20 shows the polar response of various panels;

Figure 21 shows a microphone set up for measuring the
internal pressure in the enclosure;

Figure 22 shows the internal pressure contour;

Figure 23 shows the internal pressure measured using
the array of Figure 21;

Figure 24 shows the velocity and displacement of
various panels;

Figure 25 shows the velocity spectrum of an A5 panel

in free space and enclosed;

Figure 26 shows the velocity spectrum of another A5 panel in free space and enclosed;

Figure 27 shows the power response of an A2 panel in 5 an enclosure of two depths, and

Figure 28 illustrates equalisation using filters.

In the drawings and referring more particularly to Figures 1 and 2, a sealed box loudspeaker 1 comprises a box-like enclosure 2 closed at its front by a resonant 10 panel-form acoustic radiator 5 of the kind described in WO97/09842 to define a cavity 13. The radiator 5 is energised by a vibration exciter 4 and is sealed to the enclosure round its periphery by a resilient suspension 6. The suspension 6 comprises opposed resilient strips 7, e.g. 15 of foam rubber mounted in respective L-section frame members 9,10 which are held together by fasteners 11 to form a frame 8. The interior face 14 of the back wall 3 of the enclosure 2 is formed with stiffening ribs 12 to minimise vibration of the back wall. The enclosure may be a 20 plastics moulding or a casting incorporating the stiffening ribs.

The panel in this embodiment may be of A2 size and the depth of the cavity 13 may be 90mm.

The loudspeaker embodiment of Figure 3 is generally 25 similar to that of Figures 1 and 2, but here the radiator panel 5 is mounted on a single resilient strip suspension 6, e.g. of foam rubber, interposed between the edge of the radiator 5 and the enclosure to seal the cavity. The

radiator panel size may be A5 and the cavity depth around 3
or 4 mm.

It will be appreciated that although the embodiments
of Figures 1 to 3 relate to loudspeakers, it would equally
5 be possible to produce an acoustic resonator for modifying
the acoustic behaviour of a space, e.g. a meeting room or
auditorium, using devices of the general kind of Figures 1
to 3, but which omit the vibration exciter 4.

It is shown that a panel in this form of deployment
10 can provide a very useful bandwidth with quite a small
enclosure volume with respect to the diaphragm size, as
compared with piston speakers. The mechanisms responsible
for the minimal interaction of this boundary with the
distributed mode action are examined and it is further
15 shown that in general a simple passive equalisation network
may be all that is required to produce a flat power
response. It is also demonstrated that in such a
manifestation, a DML can produce a near-ideal hemispherical
directivity pattern over its working frequency range into a
20 2Pi space.

A closed form solution is presented which is the
result of solving the bending wave equations for the
coupled system of the panel and enclosure combination. The
system acoustic impedance function is derived and is in
25 turn used to calculate the effect of the coupled enclosure
on the eigen-frequencies, and predicting the relevant
shifts and additions to the plate modes.

Finally, experimental measurement data of a number

samples of varying lump parameters and sizes are investigated and the measurements compared with the results from the analytical model.

Figure 4 illustrates a typical polar response of a
5 free DML. Note that the reduction of pressure in the plane of the panel is due to the cancellation effect of acoustic radiation at or near the edges. When a free DML is brought near a boundary, in particular parallel with the boundary surface, acoustic interference starts to take place as the
10 distance to the surface is reduced below about 15cm, for a panel of approximately 500 cm^2 surface area. The effect varies in its severity and nature with the distance to the boundary as well as the panel size. The result, nonetheless is invariably a reduction of low frequency extension,
15 peaking of response in the lower midrange region, and some aberration in the midrange and lower treble registers as shown in the example of Figure 5. Because of this, and despite the fact that the peak can easily be compensated for, application of a 'free' DML near a boundary becomes
20 rather restrictive.

When a DML is placed in a closed box or so-called "infinite baffle" of sufficiently large volume, radiation due to the rear of the panel is contained and that of the front is generally augmented in its mid and low frequency
25 response, benefiting from two aspects. First is due to the absence of interference effect, caused by the front and rear radiation, at frequencies whose acoustic wavelengths in air are comparable to the free panel dimensions; and

second, from the mid to low frequency boundary
reinforcement due to baffling and radiation into 2Pi space,
see Figure 6. Here we can see that almost 20 dB
augmentation at 100Hz is achieved from a panel of 0.25 m^2
5 surface area.

Whilst this is a definite advantage in maximising
bandwidth, it may not be possible to incorporate in
practice unless the application would lend itself to such a
solution. Suitable applications include ceiling tile
10 loudspeakers and custom in-wall installation.

In various other applications there may be a definite
advantage to utilise the benefits of the "infinite baffle"
configuration, without having the luxury of a large closed
volume of air behind the panel. Such applications may also
15 benefit from an overall thinness and lightness of the
loudspeaker. It is an object of the present invention to
bring understanding to this form of deployment and offer
analytical solutions.

A substantial volume of work supports conventional
20 piston loudspeakers in various modes of operation,
especially in predicting their low frequency behaviour when
used in an enclosure. It is noteworthy that distributed
mode loudspeakers are of very recent development and as
such there is virtually no prior knowledge of the issues
25 involved to assist with the derivation of solutions for
similar analysis. In what follows, an approach is adopted
which provides a useful set of solutions for a DML deployed
in various mechanoacoustic interface conditions including

loading with a small enclosure.

The system under analysis is shown schematically in Figure 7. In this example the front side of the panel radiates into free space, whilst the other side is loaded

5 with an enclosure. This coupled system may be treated as a network of velocities and pressures are shown in the block diagram of Figure 8. The components are, from left to right; the electromechanical driving section, the modal system of the panel, and the acoustical systems.

10 The normal velocity of the bending-wave field across a vibrating panel is responsible for its acoustic radiation. This radiation in turn leads to a reacting force which modifies the panel vibration. In the case of a DML radiating equally from both sides, the radiation impedance,

15 which is the reacting element, is normally insignificant as compared with the mechanical impedance of the panel. However, when the panel radiates into a small enclosure, the effect of acoustic impedance due to its rear radiation is no longer small, and in fact it will modify and add to

20 the scale of the modality of the panel.

This coupling, as shown in Figure 9, is equivalent to a mechanoacoustical closed loop system in which the reacting sound pressure is due to the velocity of the panel itself. This pressure modifies the modal distribution of

25 the bending wave field which in turn has an effect on the sound pressure response and directivity of the panel.

In order to calculate directivity and to inspect forces and flows within the system, it is necessary to

solve for the plate velocity. This far-field sound pressure response can then be obtained with the help of Fourier transformation of this velocity as described in an article by PANZER,J; HARRIS,N; entitled "Distributed Mode

5 Loudspeaker Radiation Simulation" presented at the 105th AES Convention, San Francisco 1998 # 4783. The forces and flows can then be found with the help of network analysis.

This problem can be approached by developing the velocities and pressures of the total system in terms of

10 the in-vacuum panel eigen-functions (3,4) as explained in CREMER,L; HECKL,M; UNGAR,E; "Structure-Borne Sound" SPRINGER 1973 and BLEVINS, R.D. "Formulas for Natural frequency and Mode Shape", KRIEGER Publ., Malabar 1984.

For example, the velocity at any point on the panel can be

15 calculated from equation (1).

$$v_{(x,y)} = \sum_{i=0}^{\infty} Y_{pi(j\omega)} \cdot F_{oi(j\omega)} \cdot \phi_{pi(x_o,y_o)} \cdot \phi_{pi(x,y)}$$

(1)

This series represents a solution to the differential

20 equation describing the plate bending waves, equation (2), when coupled to the electromechanical lumped element network as well as its immediate acoustic boundaries.

$$L_B \{ v_{(x,y)} \} - \mu \cdot \omega^2 \cdot v_{(x,y)} = j\dot{\omega} \cdot p_{m(x,y)} - j\omega \cdot p_{a(x,y)}$$

25

(2)

L_B is the bending rigidity differential operator of fourth order in x and y, v is the normal component of the bending wave velocity. μ is the mass per unit area and ω

is the driving frequency. The panel is disturbed by the mechanical driving pressure, p_m, and the acoustic reacting sound pressure field, p_a, Figure 7.

Each term of the series in equation (1) is called a

5 modal velocity, or, a "mode" in short. The model decomposition is a generalised Fourier transform whose eigen-functions Φ_{pi} share the orthogonality property with the sine and cosine functions associated with Fourier transformation. The orthogonality property of ϕ_{pi} is a

10 necessary condition to allow appropriate solutions to the differential equation (2). The set of eigen-functions and their parameters are found from the homogenous version of equation (2) i.e. after switching off the driving forces. In this case the panel can only vibrate at its natural

15 frequencies or the so-called eigen-frequencies, ϖ_i, in order to satisfy the boundary conditions.

In equation (2), $\phi_{pi(x,y)}$ is the value of the ith plate eigen-function at the position where the velocity is observed. $\phi_{pi\ (xo,yo)}$ is the eigen-function at the position

20 where the driving force $F_{pi\ (j\omega)}$ is applied to the panel. The driving force includes the transfer functions of the electromechanical components associated with the driving actuator at (x_o, y_o), as for example exciters, suspensions, etc. Since the driving force depends on the panel velocity

25 at the driving point, a similar feedback situation as with the mechanoacoustical coupling exists at the drive point(s), albeit the effect is quite small in practice.

Figure 10 gives an example of the velocity magnitude distribution of a single eigen-function across a DML panel. The black lines are the nodal lines where the velocity is zero. With increasing mode index the velocity pattern

5 becomes increasingly more complex. .For a medium sized panel approximately 200 modes must be summed in order to cover the audio range.

The modal admittance, $Y_{pi(j\omega)}$, is the weighting function of the modes and determines with which amplitude and in

10 which phase the ith mode takes part in the sum of equation (1). Y_{pi}, as described in equation (3), depends on the driving frequency, the plate eigen-value and, most important in the context of this paper, on the acoustic impedance of the enclosure together with the impedance due

15 to the free field radiation.

$$Y_{pi(s)} = \frac{1}{R_{pi}} \cdot \frac{s_p \cdot d_{pi}}{s_p^2 + s_p \cdot d_{pi} + \gamma_{pi\,v}^2}$$

(3)

$s_p = s/\omega_p$ is the Laplace frequency variable normalised to

20 the fundamental panel frequency, ω_p, which in turn depends on the bending stiffness K_p and mass M_p of the panel, namely $\omega_p^2 = K_p/M_p$. R_{pi} is the modal resistance due to material losses and describes the value of $Y_{pi(j\omega)}$ at resonance when s_p = λ_{pi}. λ_{pi} is a scaling factor and is a function of the ith

25 plate eigen-value λ_{pi} and the total radiation impedance Z_{mai} as described in equation (4).

$$Y_{pi(s)} = \sqrt{\lambda_{pl}^4 + s_p \cdot Z_{mai(j\omega)}} \cdot \sqrt{\frac{1}{K_p \cdot M_p}}$$

(4)

In the vacuum case ($Z_{mai}=0$) the second term in equation

(3) becomes a band-pass transfer function of second order

with damping factor d_{pi}. Figure 11 shows the magnitudes of

the frequency response of the in-vacuum $Y_{pi(j\omega)}$ for the first

ten modes of a panel, when clamped at the edges. The panel

eigen-frequencies coincide with the peaks of these curves.

If the same panel is now mounted onto an enclosure,

the modes will not only be shifted in frequency but also

modified, as seen in Figure 12. This happens as a result

of the interaction between the two modal systems of the

panel and the enclosure, where the modal admittance of the

total system is no longer a second order function as in the

in-vacuum case. In fact, the denominator of equation (3)

could be expanded in a polynomial of high order, which will

reflect the resulting extended characteristic function.

The frequency response graphs of Figure 13 shows the

effect of the enclosure on the panel velocity spectrum.

The two frequency response curves are calculated under

identical drive condition, however, the left-hand graph

displays the in-vacuum case, whilst the right hand graph

shows the velocity when both sides of the panel are loaded

with an enclosure. A double enclosure was used in this

example in order to exclude the radiation impedance of air.

The observation point is at the drive point of the exciter.

Clearly visible is the effect of the panel eigen-frequency shift to higher frequencies in the right diagram, which was also seen in Figure 12. It is noteworthy that as a result of the enclosure influence, and the subsequent increase in

5 the number and density of modes, a more evenly distributed curve describing the velocity spectrum is obtained.

The mechanical radiation impedance is the ratio of the reacting force, due to radiation, and the panel velocity. For a single mode, the radiation impedance can be regarded

10 as constant across the panel area and may be expressed in terms of the acoustical radiated power P_{ai} of a single mode. Thus the modal radiation impedance of the ith mode may be described by equation (5).

15

$$Z_{mai} = 2 \cdot \frac{P_{ai}}{<v_i>^2}$$

(5)

$<v_i>$ is the mean velocity across the panel associated with the ith mode. Since this value is squared and therefore always positive and real, the properties of the

20 radiation impedance Z_{mai} are directly related to the properties of the acoustical power, which is in general a complex value. The real part of P_{ai} is equal to the radiated far-field power, which contributes to the resistive part of Z_{mai}, causing damping of the velocity field of the panel.

25 The imaginary part of P_{ai} is caused by energy storing mechanisms of the coupled system, yielding to a positive or negative value for the reactance of Z_{mai}.

A positive reactance is caused by the presence of an

acoustical mass. This is typical, for example, of radiation into free space. A negative reactance of Z_{mai}, on the other hand, is indicative of the presence of a sealed enclosure with its equivalent stiffness. In physical
5 terms, a 'mass' type radiation impedance is caused by a movement of air without compression, whereas a 'spring' type impedance exists when air is compressed without shifting it.

The principal effect of the imaginary part of the
10 radiation impedance is a shift of the in-vacuum eigen-frequencies of the panel. A positive reactance of Z_{mai} (mass) causes a down-shift of the plate eigen-frequencies, whereas a negative reactance (stiffness) shifts the eigen-frequencies up. At a given frequency, the pane-mode itself
15 dictates which effect will be dominating. This phenomenon is clarified by the diagram of Figure 14, which shows that symmetrical mode shapes cause compression of air, 'spring' behaviour, whereas asymmetrical mode shapes shift the air side to side, yielding an acoustical 'mass' behaviour. New
20 modes, which are not present in either system when they are apart, are created by the interaction of the panel and enclosure reactances.

Figure 15 shows the frequency response of the imaginary part of the enclosure radiation impedance. The
25 left-hand graph displays a 'spring-type' reactance, typically produced by a symmetrical panel-mode. Up to the first enclosure eigen-frequency the reactance is mostly negative. In-vacuum eigen-frequencies of the panel, which

are within this frequency region, are shifted up. In contrast the right diagram displays a 'mass-type' reactance behaviour, typically produced by an asymmetrical panel mode.

5   If the enclosure is sealed and has a rigid wall parallel to the panel surface, as in our case here, then the mechanical radiation impedance for the ith-plate mode is (5):

$$Z_{mai} = -j \cdot \omega \cdot \rho_a \cdot \frac{A_0^2}{A_d} \cdot \sum_{k,l} \frac{\Psi_{(i,k,l)}^2}{k_{z(k,l)} \cdot \tan(k_{z(k,l)} \cdot L_{dz})}$$

10

(6)

$\psi_{(i,k,l)}$ is the coupling integral which takes into account the cross-sectional boundary conditions and involves the plate and enclosure eigen-functions. The
15 index, i, in equation (6) is the plate mode-number; L_{dz} is the depth of the enclosure; and k_z is the modal wave-number component in the z-direction (normal to the panel). For a rigid rectangular enclosure k_z is described by equation (7):

20

$$k_{z(k,l)} = \sqrt{k_a^2 - \left[\left(\frac{k \cdot \pi}{L_{dx}}\right)^2 + \left(\frac{l \cdot \pi}{L_{dy}}\right)^2\right]}$$

(7)

The indices, k and l, are the enclosure cross-mode numbers in x and y direction, where L_{dx} and L_{dy} are enclosure
25 dimensions in this plane. A_0 is the area of the panel and A_d is cross-sectional area of the enclosure in the x and y plane.

Equation (6) is a complicated function, which describes the interaction of the panel modes and the enclosure modes in detail. In order to understand the nature of this formula, let us simplify it by constraining the system to the first mode of the panel and to the z-modes of the enclosure only (k=l=0). This will result in the following simplified relationship.

$$Z_{ma0} = -j \cdot Z_a \cdot \frac{A_0^2}{A_d} \cdot \cot(k_z \cdot L_{dz})$$

(8)

Equation (8) is the well known driving point impedance of a closed duct (6). If the product $k_z.L_{dz} \ll 1$ then a further simplification can be made as follows.

$$Z_{ma0} = A_0^2 \cdot \frac{1}{j \cdot \omega \cdot C_{ab}}$$

(9)

where $C_{ab} = \dot{V}_b/(\rho_a.c_a^2)$ is the acoustical compliance of the enclosure of volume V_b. Equation (9) is the low frequency lumped element model of the enclosure. If the source is a rigid piston of mass M_{ms} with a suspension having a compliance C_{ms} then the fundamental 'mode' has the eigen-value $\lambda_{po} = 1$ and the scaling factor of the coupled system of equation (4) becomes the well known relationship as shown in equation (10),[1].

$$\gamma_{po} = \sqrt{1 + \frac{C_{ms}}{C_{mb}}}$$

(10)

with the equivalent mechanical compliance of the enclosure air volume $C_{mb} = C_{ab}/A_0^2$.

Various tests were carried out to investigate the effect of a shallow back enclosure on DM loudspeakers. In addition to bringing general insight into the behaviour of DNM panels in an enclosure, the experiments were designed to help verify the theoretical model and establish the extent to which such models are accurate in predicting the behaviour of the coupled modal system of a DML panel and its enclosure.

Two DML panels of different size and bulk properties were selected as our test objects. It was decided that these would be of sufficiently different size on the one hand, and of a useful difference in their bulk properties on the other, to cover a good range in scale. The first set 'A' was selected as a small A5 size panel of 149mm x 210mm with three different bulk mechanical properties. These were A5-1, polycarbonate skin on polycarbonate honeycomb; A5-2 carbon fibre on Rohacell; and A5-3, Rohacell without skin. Set 'B' was chosen to be eight times larger, approximately to A2 size of 420mm x 592mm. A2-1 was constructed with glass fibre skin on polycarbonate honeycomb core, whilst A2-2 was carbon fibre skin on aluminium honeycomb. Table 1 lists the bulk properties of these objects. Actuation was achieved by a single electrodynamic moving coil exciter at the optimum position. Two exciter types were used, where they suited most the

size of the panels under test. In the case of A2 panels a

25mm exciter was employed with Bl = 2.3 Tm, Re = 3.7 Ω and

Le - 60 μH, whilst a 13mm model was used in the case of the

smaller A5 panels with Bl - 1.0 Tm, Re=7.3 Ω and Le=36 μH.

Panel	Type	B (Nm)	μ (Kg/m^2)	Zm (Ns/m)	Size (mm)
A2-1	Glass on PC Core	10.4	0.89	24.3	5 x 592 x 420
A2-2	Carbon on AI Core	57.6	1.00	60.0	7.2 x 592 x 420
A5-1	PC on PC core	1.39	0.64	7.5	2 x 210 x 149
A5-2	Carbon on Rohacell	3.33	0.65	11.8	2 x 210 x 149
A5-3	Rohacell core	0.33	0.32	2.7	3 x 210 x 149

5

    Panels were mounted onto a back enclosure with

adjustable depth using a soft polyurethane foam for

suspension and acoustic seal. The enclosure depth was made

adjustable on 16,28,40 and 53mm for set 'A' and on 20,50,95

10 and 130mm for set 'B' panels. Various measurements were

carried out at different enclosure depths for every test

case and result documented.

    Panel velocity and displacement were measured using a

Laser Vibrometer. The frequency range of interest was

15 covered with a linear frequency scale of 1600 points. The

set-up shown in Figure 16 was used to measure the panel

mechanical impedance by calculating the ratio of the

applied force to the panel velocity at the drive point.

20

$$Z_m = \frac{F}{V}$$

    In this procedure, the applied force was calculated

from the lump parameter information of the exciter. Although panel velocity in itself feeds back into the electromechanical circuit, its coupling is quite weak. It can be shown that for small values of exciter Bl,(1-3 Tm),

5 providing that the driving amplifier output impedance is low (constant voltage), the modal coupling back to the electromechanical system is sufficiently weak to make this assumption plausible. Small error arising from this approximation was therefore ignored. Figures 18a to f show

10 the mechanical impedance of the A5-1 and A5-2 panels, derived from the measurement of panel velocity and the applied force measured by the Laser Vibrometer. Note that the impedance minima for each enclosure depth occur at the system resonance mode.

15   Sound pressure level and polar response of the various panels were measured in a large space of 350 cubic metres and gated at 12 to 14ms for anechoic response using MLSSA, depending on the measurement. Power measurements were carried out employing a 9-microphone array system, as shown

20 in Figure 17d and in a set-up shown in Figure 17a. These are plotted in Figures 19a to f for various enclosure depths. System resonance is highlighted by markers on the graphs.

Polar response of the A5-1 and A5-2 panels were

25 measured for a 28mm deep enclosure and the result is shown in Figures 20a and b. When compared with the polar plot of the free DML in Figure 1, they demonstrate the significance of the closed-back DML in its improved directivity.

To investigate further the nature and the effect of enclosure on the panel behaviour, especially at the combined system resonance, a special jig was made to allow the measurement of the internal pressure of the enclosure

5 at nine predetermined points as shown in Figure 21. The microphone was inserted in the holes provided within the back-plate of an A5 enclosure jig at a predetermined depth, while the other eight position holes were tightly blocked with hard rubber grommets. The microphone was mechanically

10 isolated from the enclosure by an appropriate rubber grommet during the measurement.

From this data, a contour plot was created to show the pressure distribution at system resonance and that either side of this frequency as shown in Figures 22a to c. The

15 pressure frequency response was also plotted for the nine positions as shown in Figure 27. This graph exhibits good definition in the region of resonance for all curves associated with the measurement points within the enclosure. However, the pressure tends to vary across the

20 enclosure cross-sectional area as the frequency is increased.

The normal component of velocity and displacement across the panels was measured with a Scanning Laser Vibrometer. The velocity and displacement distribution

25 across the panels were plotted to investigate the behaviour of the panel around the coupled system resonance. The results were documented and a number of the cases are shown in Figures 24a to d. These results suggest a timpanic

modal behaviour of the panel at resonance, with the whole of the panel moving, albeit at a lesser velocity and displacement as one moves towards the panel edges.

In practice this behaviour is consistent for all boundary conditions of the panel, although the mode shape will vary from case to case depending on a complex set of parameters, including panel stiffness, mass, size and boundary conditions. In the limit and for an infinitely rigid panel, this system resonance will be seen as the fundamental rigid body mode of the piston acting on the stiffness of the enclosure air volume. It was found to be convenient to call the DML system resonance, the 'Whole Body Mode' or WBM.

The full theoretical derivations of the coupled system has been implemented in a suite of software by New Transducers Limited. A version of this package was used to simulate the mechanoacoustical behaviour of our test objects in this paper. This package is able to take into account all the electrical, mechanical and acoustical variables associated with a panel, exciter(s) and mechanoacoustical interfaces with a frame or an enclosure and predict, amongst other parameters, the far-field acoustic pressure, power and directivity of the total system.

Figure 25a shows the log-velocity spectrum of a free radiating, A5-1 panel clamped in a frame, radiating in free space equally from both sides. The solid line represents the simulation curve and the dashed line is the measure

24

velocity spectrum. At low frequencies the panel goes in resonance with the exciter. The discrepancy in the frequency range above 1000 Hz is due to the absence of the free field radiation impedance in the simulation model.

5   Figure 25b shows the same panel as in Figure 25a but this time loaded with two identical enclosures, one on each side of the panel, with the same cross-section as the panel and a depth of 24mm. A double enclosure was designed and used in order to exclude the radiation impedance of free

10 field on one side of the panel and make the experiment independent of the free field radiation impedance. It is important to note that this laboratory set-up was used for theory verification only.

In order to enable velocity measurement of the panel,

15 the back walls of the two enclosures were made from a transparent material to allow access by the laser beam to the panel surface. This test was repeated using panel A5-3 Rohacell without skin, with different bulk properties and the result is shown in Figures 26a and b. In both cases

20 simulation was performed using 200 point logarithmic range, whilst the laser measurement used 1600 point linear range.

From the foregoing theory and work, it is clear that a small enclosure fitted to a DML will bring with it, amongst a number of benefits, a singular drawback. This manifests

25 itself in an excess of power due to WBM at the system resonance as shown in Figures 27a and b. It is noteworthy that apart from this peak, in all other aspect the enclosed DML can offer a substantially improved performance

including increased power bandwidth.

It has been found that in most cases a simple second order band-stop equalisation network of appropriate Q matching that of the power response peak, may be designed 5 to equalise the response peak. Furthermore in some cases a single pole high-pass filter would often adjust for this by tilting the LF region, to provide a broadly flat power response. Due to the unique nature of DML panels and their resistive electrical impedance response, whether the 10 filter is active or passive, its design will remain very simple. Figure 28a shows where a band-stop passive filter has been incorporated for equalisation. Further examples may be seen in Figures 28b and c that show simple pole EQ with a capacitor used in series with the loudspeakers.

15   When a free DML is used near and parallel to a wall, special care must be taken to ensure minimal interaction with the latter, due to its unique complex dipolar characteristics. This interaction is a function of the distance to the boundary, and therefore, cannot be 20 universally fixed. Full baffling of the panel has definite advantages in extending the low frequency response of the system, but this may not be a practical proposition in a large number of applications.

A very small enclosure used with a DML will render it 25 independent of its immediate environment and make the system predictable in its acoustical performance. The mathematical model developed demonstrates the level of complexity for a DML in the coupled system. This throws a

sharp contrast between the prediction and design of a DML and that of the conventional piston radiator. Whilst the mechanoacoustical properties of a cone-in-box may be found by relatively simply calculations (even by a hand
5 calculator) those associated with a DML and its enclosure are subject to complex interactive relationships which render this system impossible to predict without the proper tools.

The change in system performance with varying
10 enclosure volume is quite marked in the case where the depth is small compared with the panel dimensions. However, it is also seen that beyond a certain depth the increase in LF response become marginal. This of course is consistent with behaviour of a rigid piston in an enclosure. As an
15 example, an A2 size panel with 50mm enclosure depth can be designed to have a bandwidth extending down to about 120Hz, Figure 24.

Another feature of a DML with a small enclosure is seen to be a significant improvement in the mid and high
20 frequency response of the system. This is in many of the measured and simulated graphs in this paper and of course anticipated by the theory. It is clear that the increase in the panel system modality is mostly responsible for this improvement, however, enclosures losses might also
25 influence this by increasing the overall damping of the system.

As a natural consequence of containing the rear radiation of the panel, the directivity of the enclosed
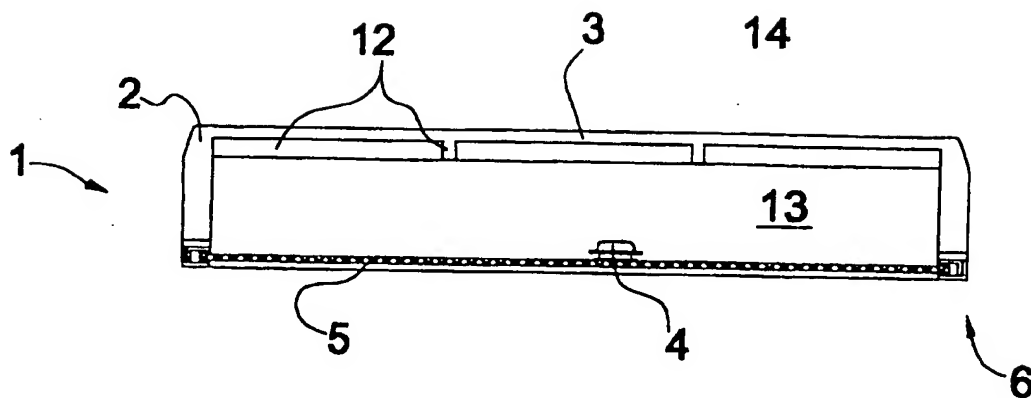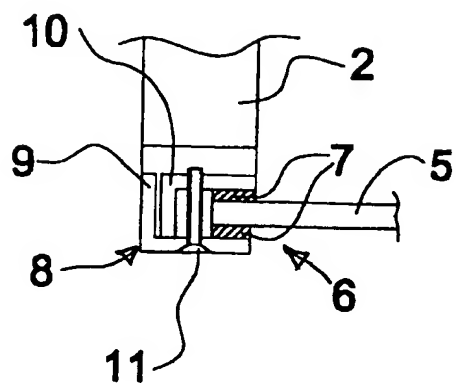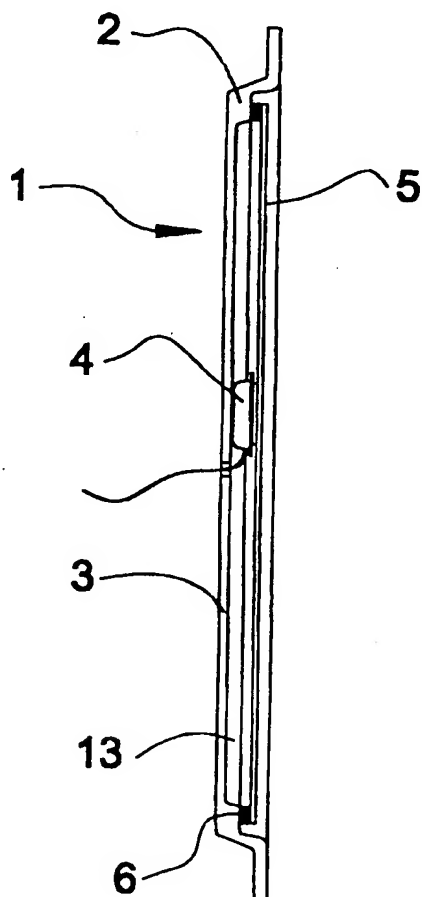
system changes substantially from a dipolar shape to a near
cardioid behaviour as shown in Figure 17. It is envisaged
that the directivity associated with a closed-back DML may
find use in certain applications where stronger lateral
5 coverage is desirable.

Power response measurements were found to be most
useful when working with the enclosed DM system, in order
to observe the excessive energy region that may need
compensation. This is in line with other work done on DM
10 loudspeakers, in which it has been found that the power
response is the most representative acoustic measurement
correlating well to the subjective performance of a DML.
Using the power response, it was found that in practice a
simple band-pass or a single pole high-pass filter is all
15 that is needed to equalise the power response in this
region.

CLAIMS

1.   An acoustic device comprising a resonant multi-mode acoustic panel having opposed faces, means defining a cavity enclosing at least a portion of one panel face and arranged to contain acoustic radiation from the said portion of the panel face, wherein the cavity is such as to modify the modal behaviour of the panel.

2.   An acoustic device according to claim 1, wherein the cavity size is such as to modify the modal behaviour of the panel.

3.   An acoustic device according to claim 2, wherein the cavity is shallow.

4.   An acoustic device according to claim 3, wherein the cavity is sufficiently shallow that the rear face of the cavity facing the said one panel face causes fluid coupling to the panel.

5.   An acoustic device according to claim 4, wherein X and Y cross modes are generally dominant.

6.   An acoustic device according to any preceding claim, wherein the cavity is sealed.

7.   An acoustic device according to any preceding claim, wherein the ratio of the cavity volume to panel area $(ml:cm^2)$ is in the range about 10:1 to 0.2:1.

8.   An acoustic device according to any preceding claim, wherein the panel is mounted in and sealed to the cavity defining means by a peripheral surround.

9.   An acoustic device according to claim 8, wherein the surround is resilient.

10. A loudspeaker comprising an acoustic device as claimed in any preceding claim, and having a vibration exciter arranged to apply bending wave vibration to the resonant panel to produce an acoustic output.

5 11. A method of multiplying the modal behaviour of a resonant panel acoustic device, comprising bringing the resonant panel into close proximity with a boundary surface to define a resonant cavity therebetween.
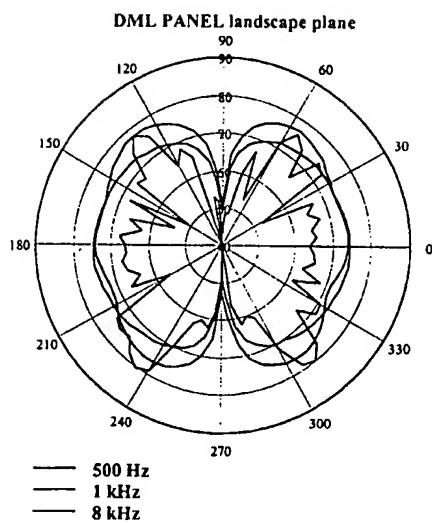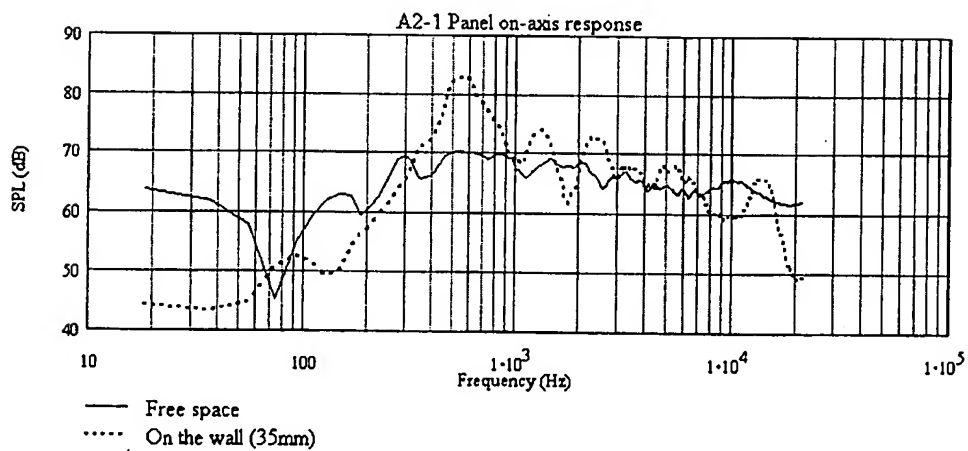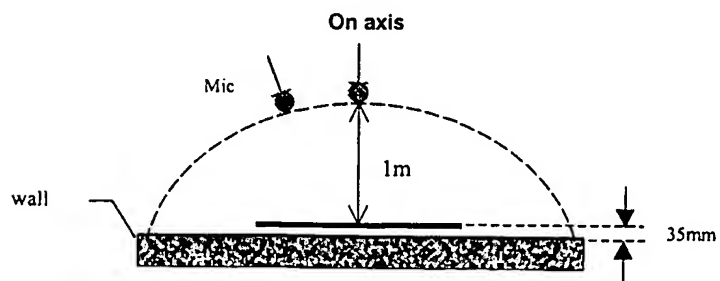
Figure 1.

Figure 2.

Figure 3.

DML PANEL landscape plane

——— 500 Hz
——— 1 kHz
——— 8 kHz

Fig. 4

A2-1 Panel on-axis response

——— Free space
····· On the wall (35mm)

Fig. 5

Fig. 17a

Fig. 17b

····· Free Panel
——— in IB (Ceiling Tile)

Fig. 6

Fig. 7

Network of mechanical components

U

Pressure p_m and velocity v at the drive point

Panel

Radiation impedance of free field

Radiation impedance of enclosure

Fig. 8

Driving pressure p_m

+

Bending wave field of the Panel

Velocity

Sound pressure p_a

Radiation into Free space and enclosure

Fig. 9

Fig. 10

Modal admittance term

200   500   1k   Frequency   2k

Fig. 11

Modal admittance term

5S

0

200   500   1k   Frequency   2k

Fig. 12

Fig. 13

Fig. 14

ohm   Spring type radiation reactance

200
160
120
80
40
0
-40
-80
-120
-160
-200

        50   100   1k   5k

ohm   Mass type radiation reactance

200
160
120
80
40
0
-40
-80
-120
-160
-200

        50   100   1k   5k

Fig. 15

| Laser Vibrometer Vp | PANEL System Zm |
| Power Amp Zo | EXCITER Bl, Re, Le |

F ↑   V ↓
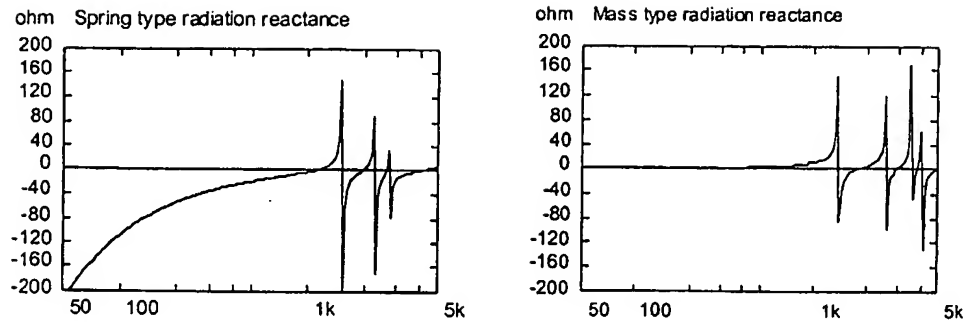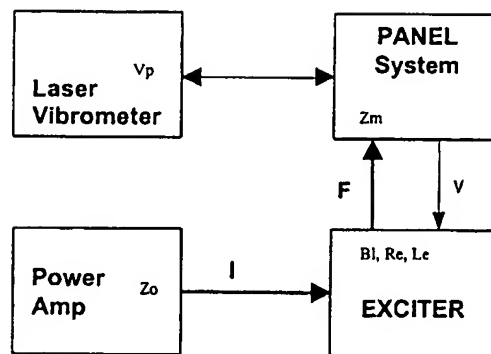
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Fig. 16
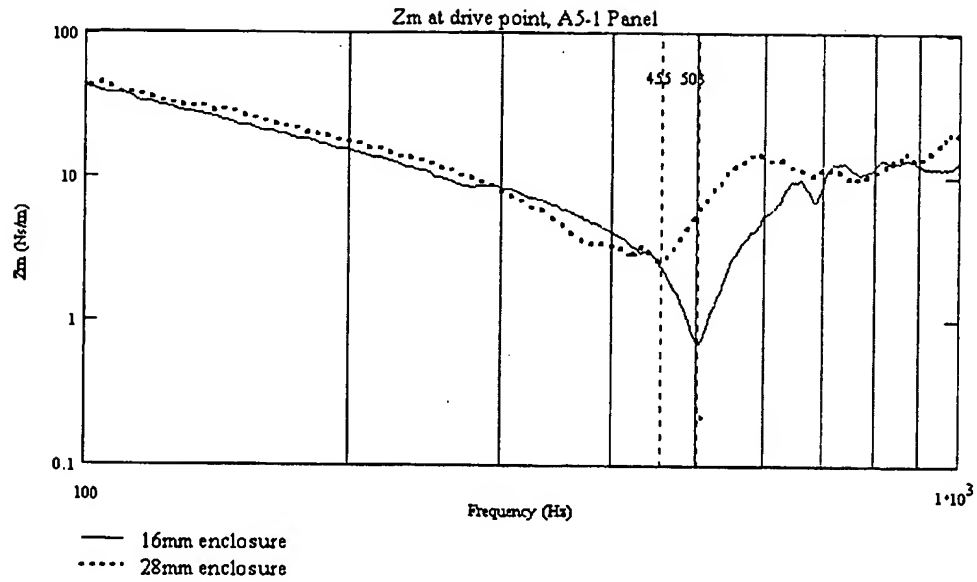
Fig. 18a A5-1 Panel

Fig. 18b A5-1 Panel
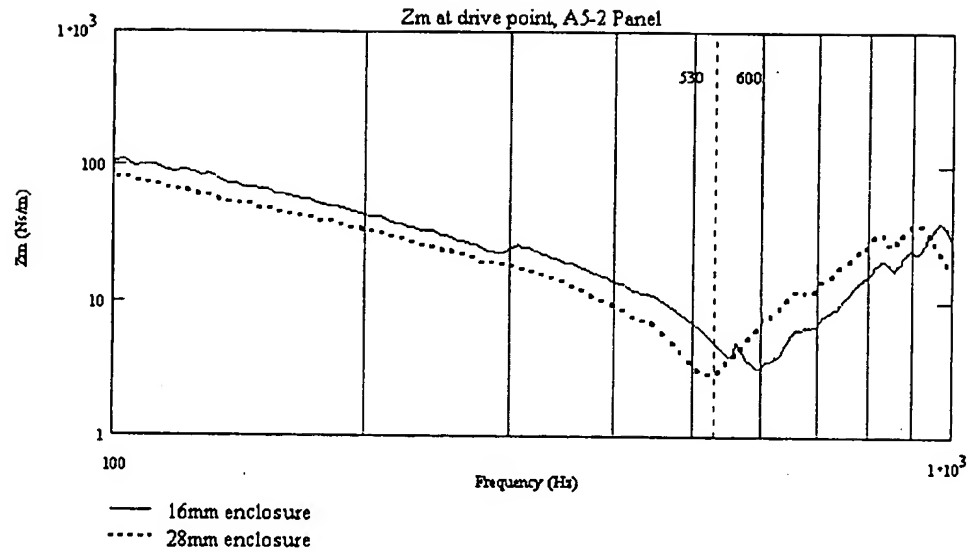
Fig. 18c A5-2 Panel
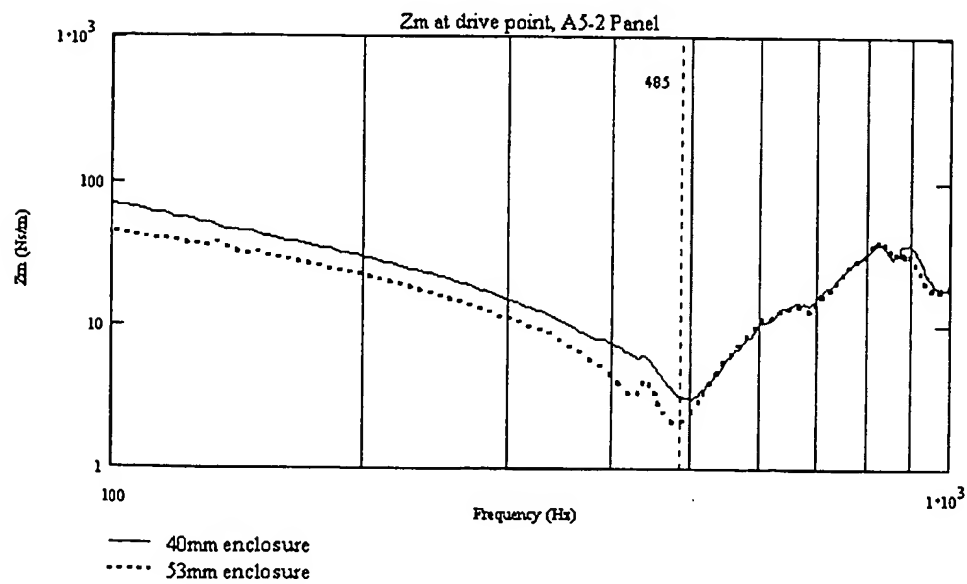
Fig. 18d A5-2 Panel
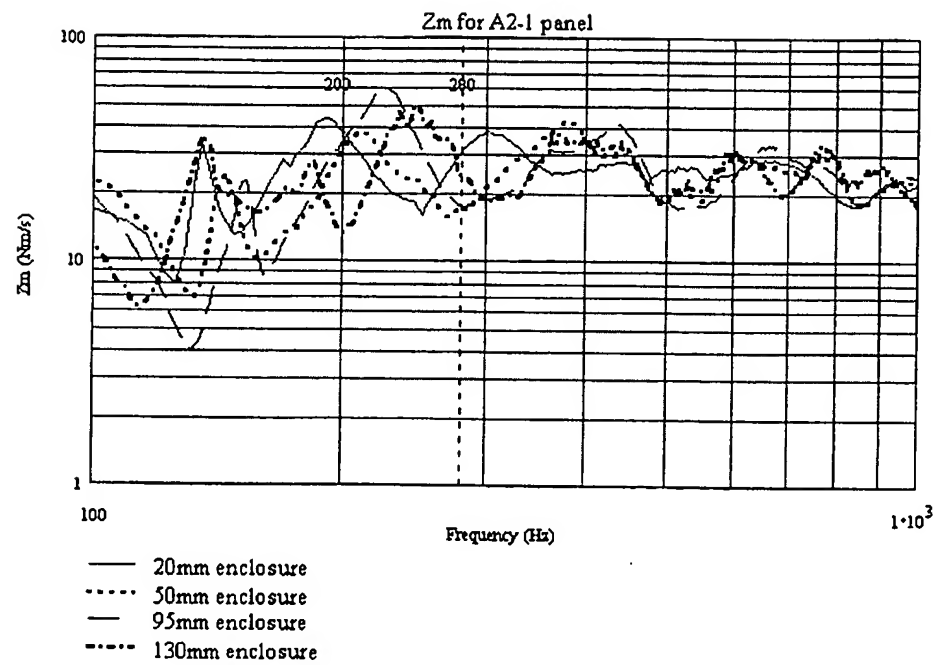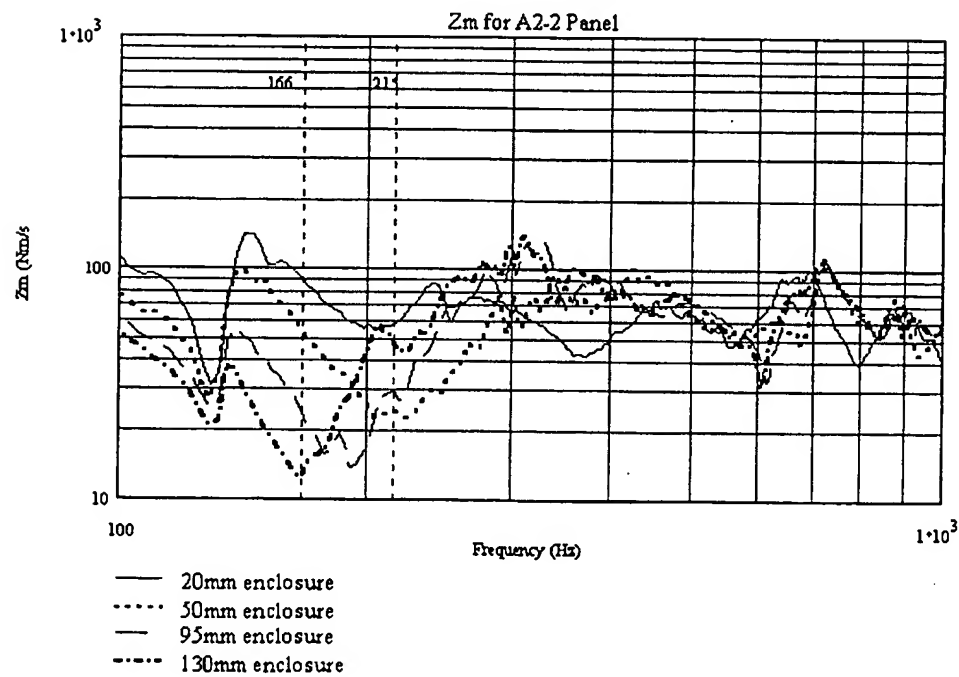
Fig. 18e. A2-1 Panel

Fig. 18f A2-2 Panel
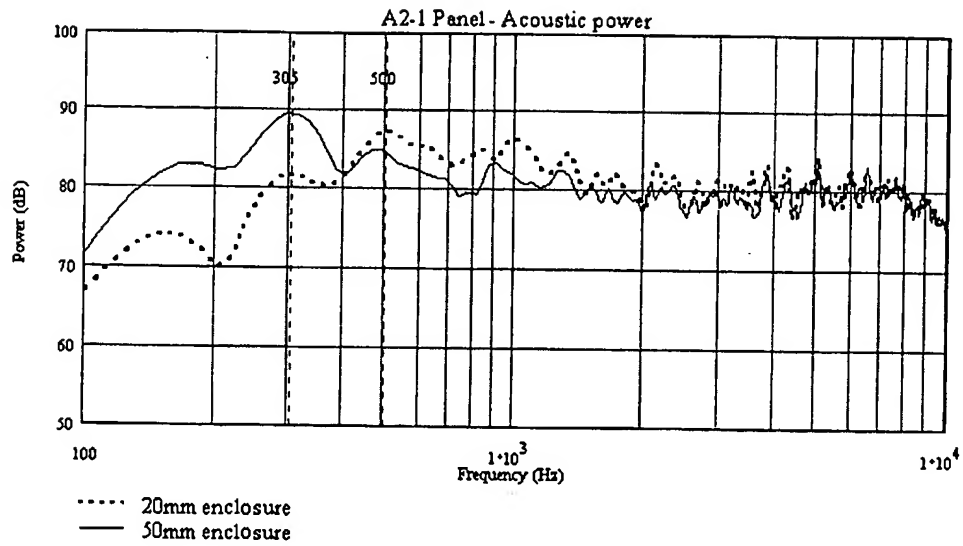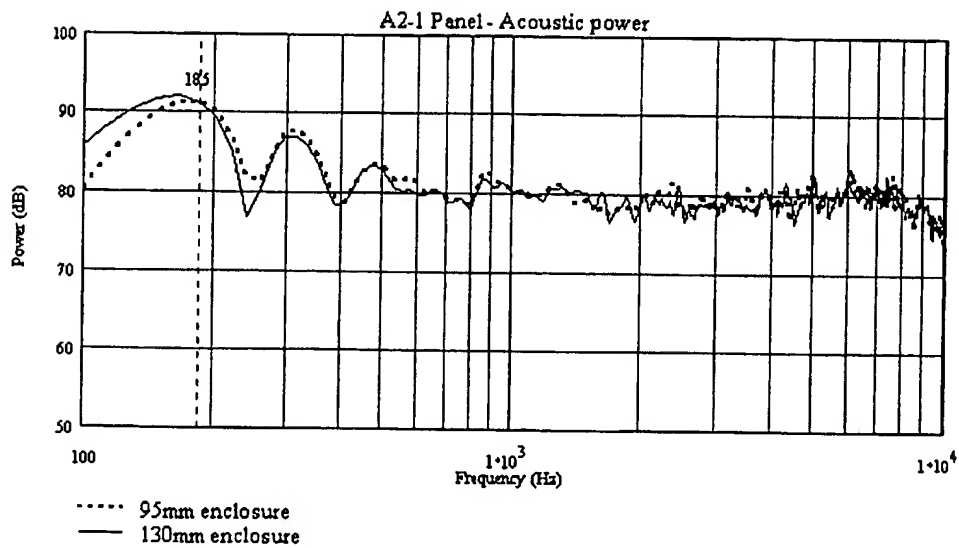
Fig. 19a A2-1 Panel

Fig. 19b A2-2 Panel
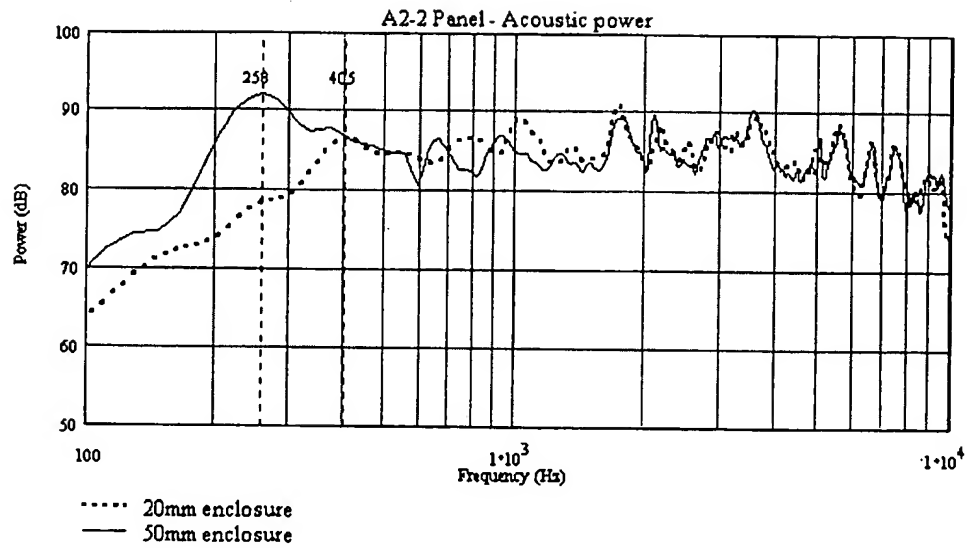
Fig. 19c A2-1 Panel

Fig. 19d A2-1 Panel

A5-1

— 1 kHz
.... 3 kHz
---- 8 kHz

Fig. 20a A5-1 Panel

A5-2

— 1.7 kHz
.... 3 kHz
---- 8 kHz
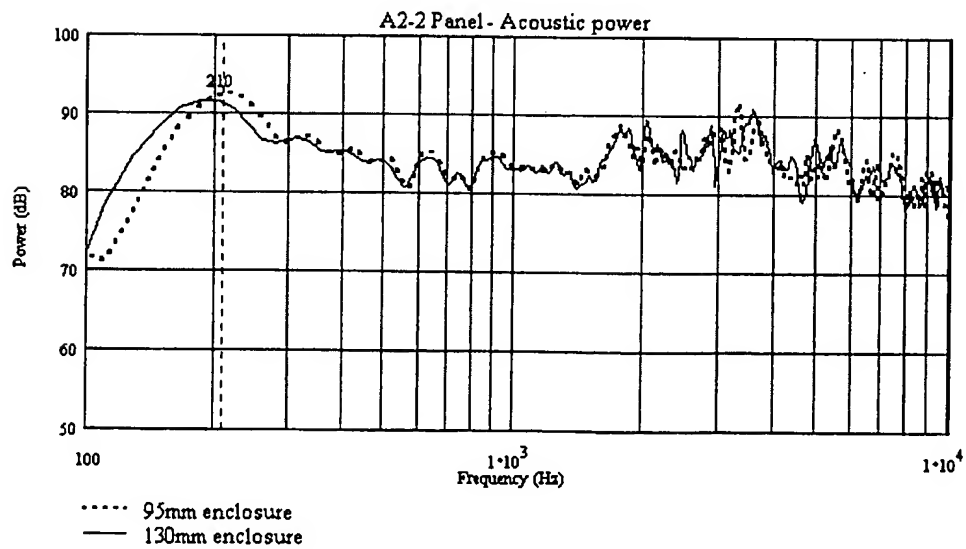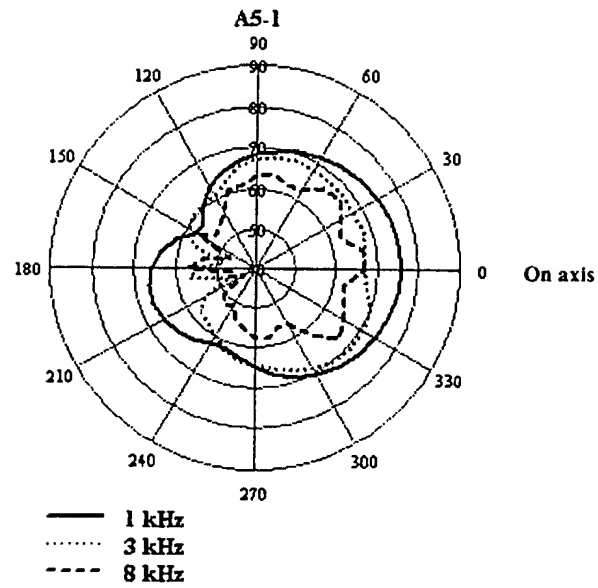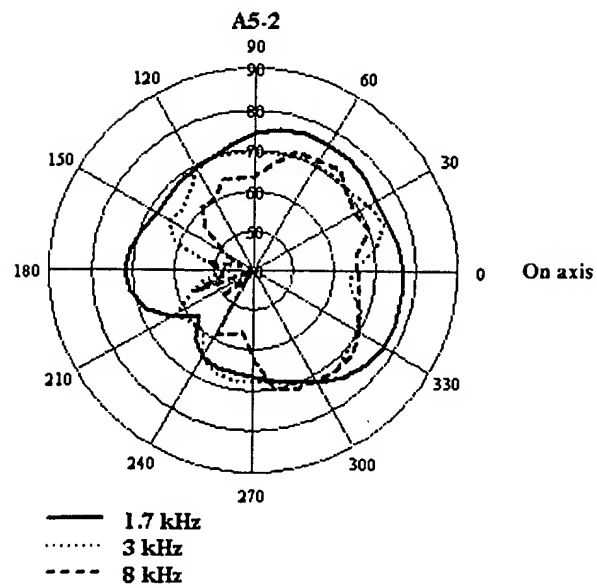
Fig. 20b A5-2 Panel
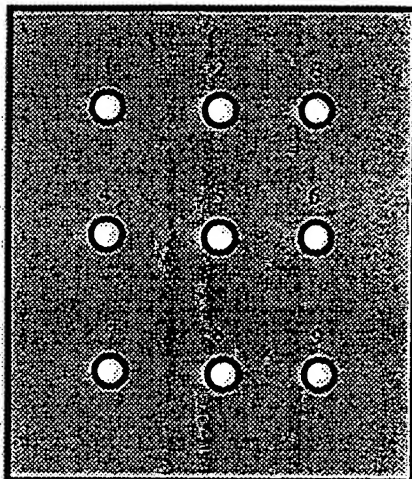
Fig. 21
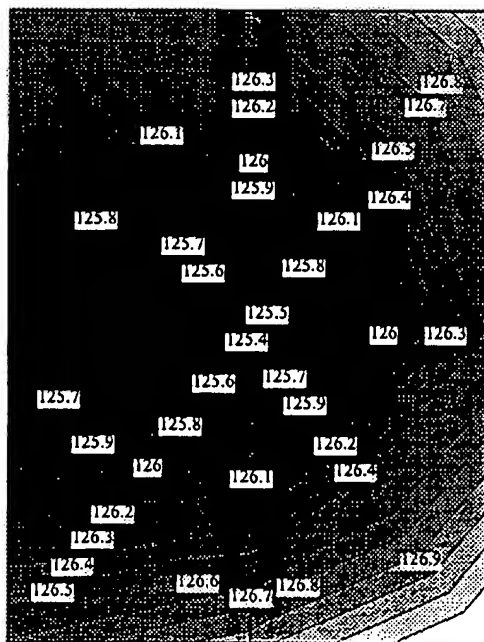
Fig. 22a A5-1 Panel 483Hz
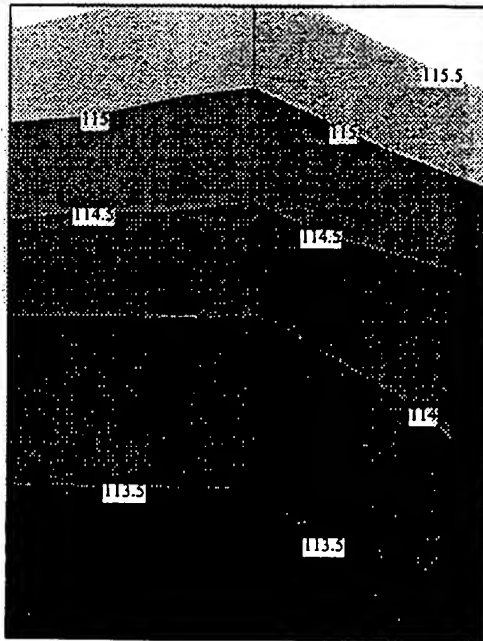
Fig. 22b A5-1 panel 301Hz
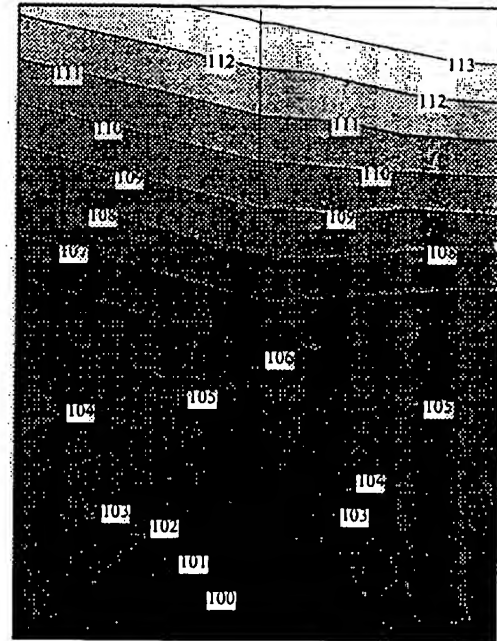
Fig.22c A5-1 panel 817Hz
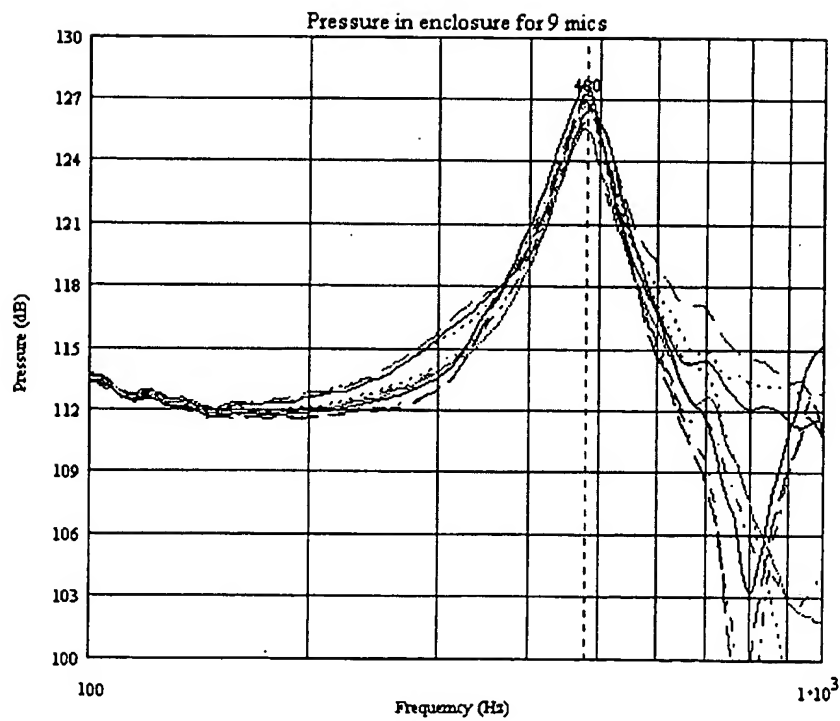
Fig. 23 A5-1 panel

A2-1 Panel velocity at 165 Hz.
50mm Enclosure
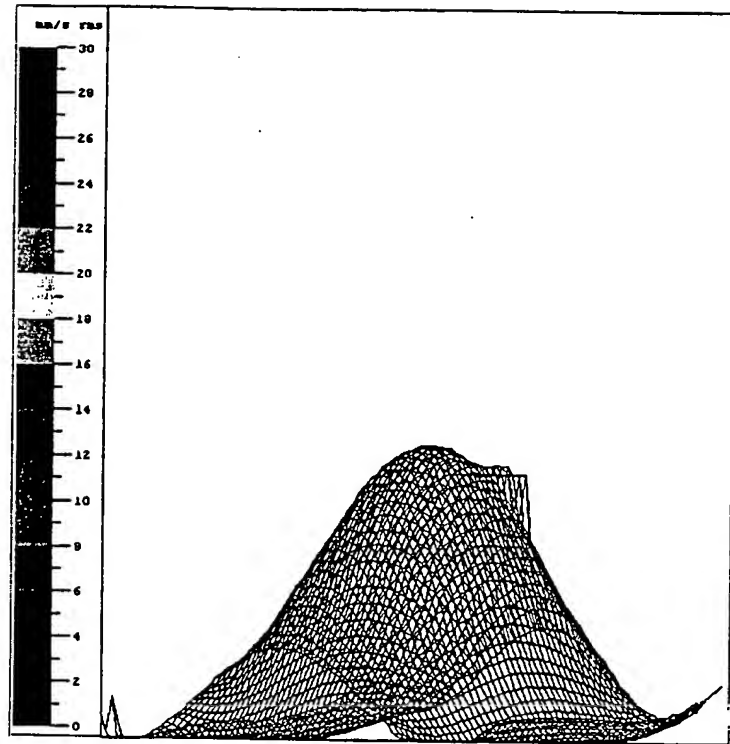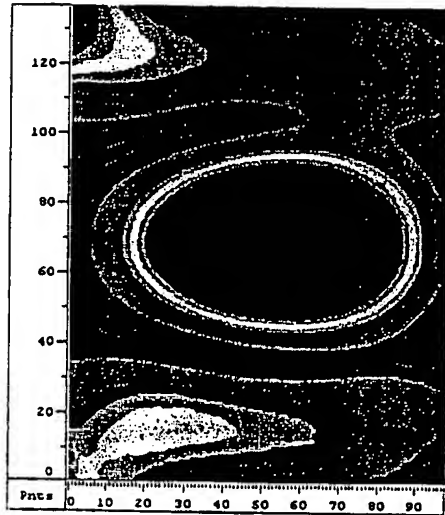
Fig. 24a

A2-1 Panel velocity at 153 Hz.
(95mm Enclosure)
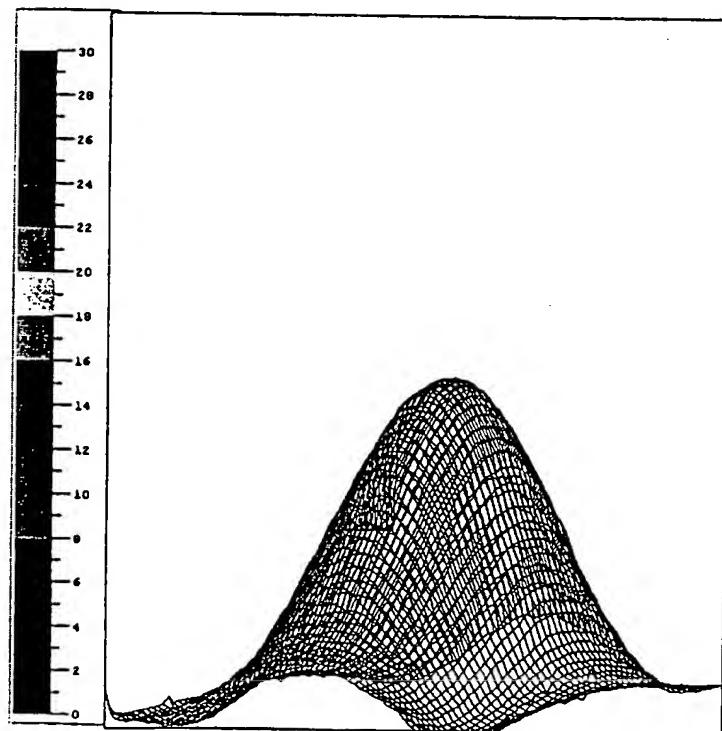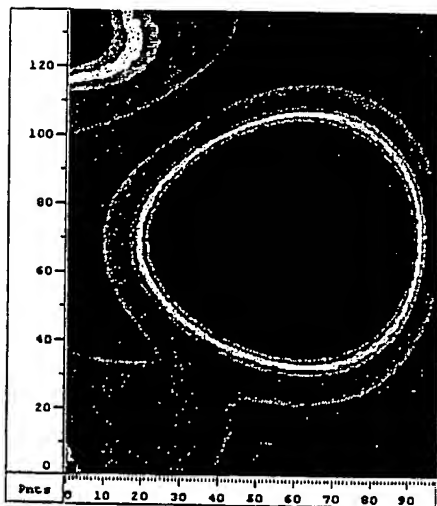
Fig. 24b

A2-2 Panel velocity at 194 Hz.
(95mm Enclosure)
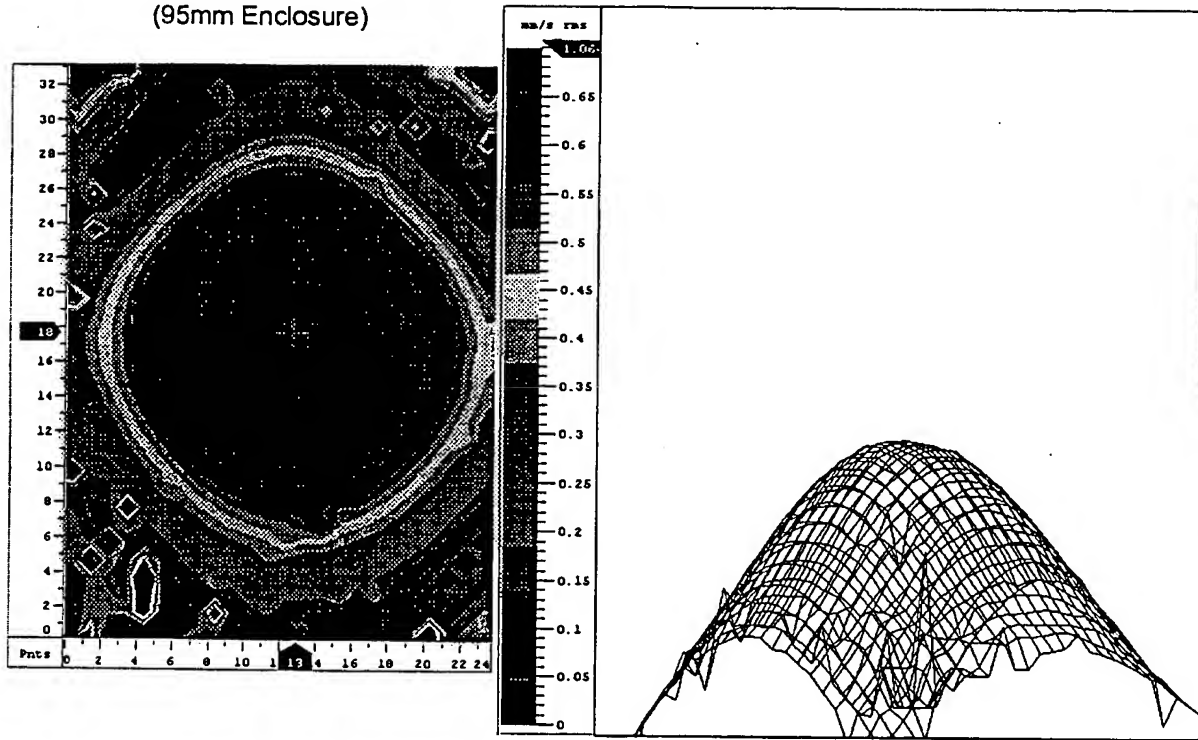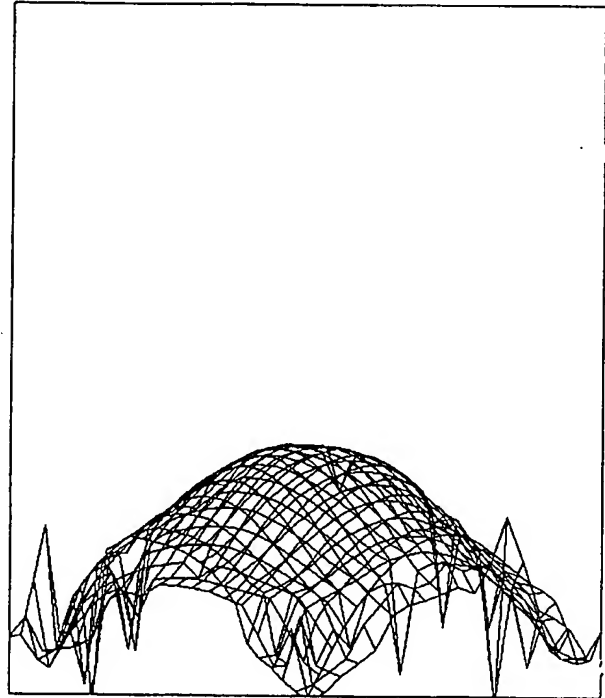
Fig. 24c

Fig. 24d

A2-2 Panel velocity at 194 Hz.
(95mm Enclosure)
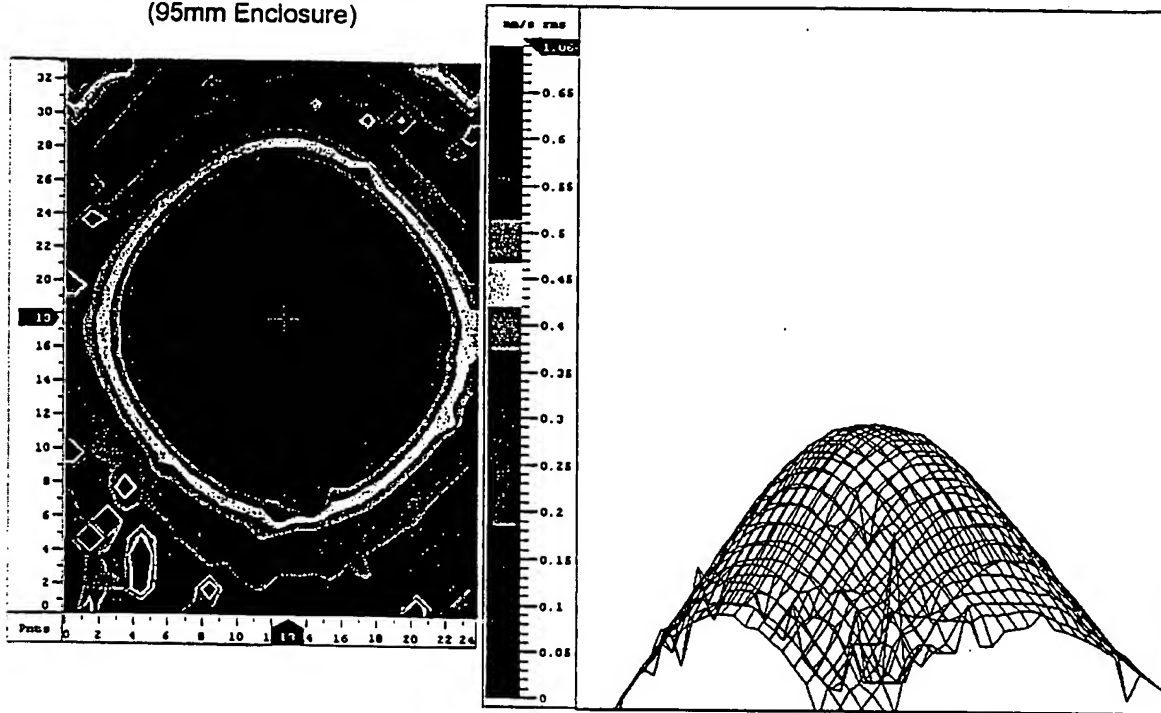
Fig. 24c

A2-2 Panel velocity at 166 Hz.
(130mm Enclosure)
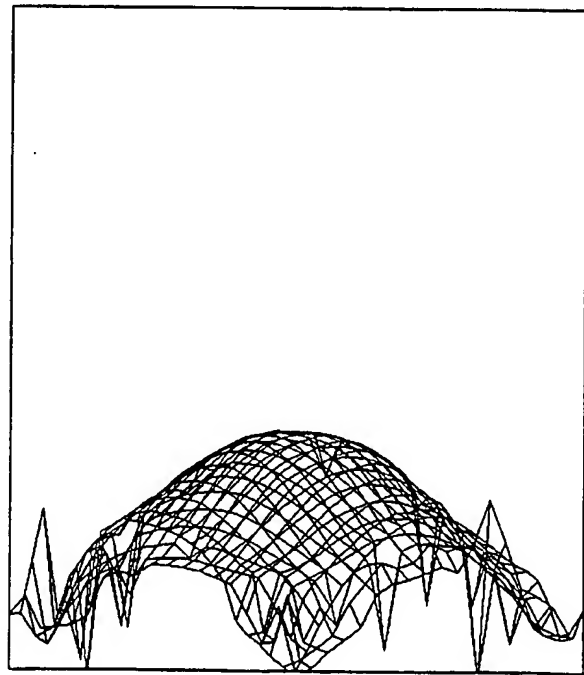
Fig. 24d
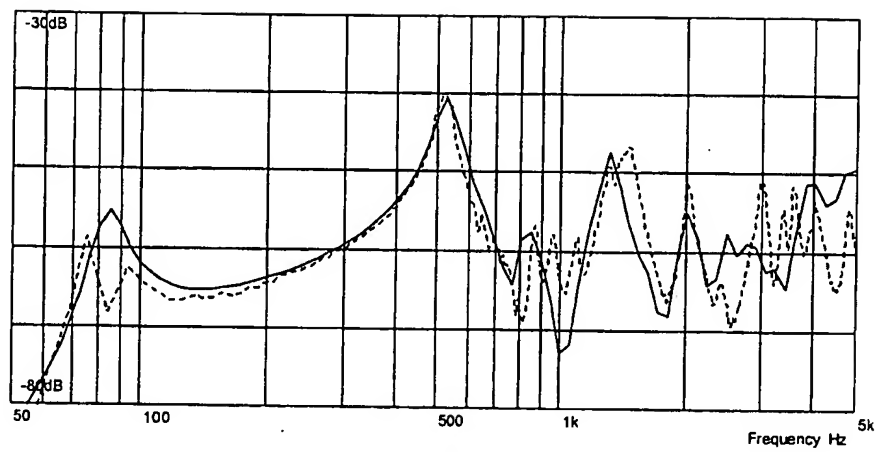
Fig. 25a
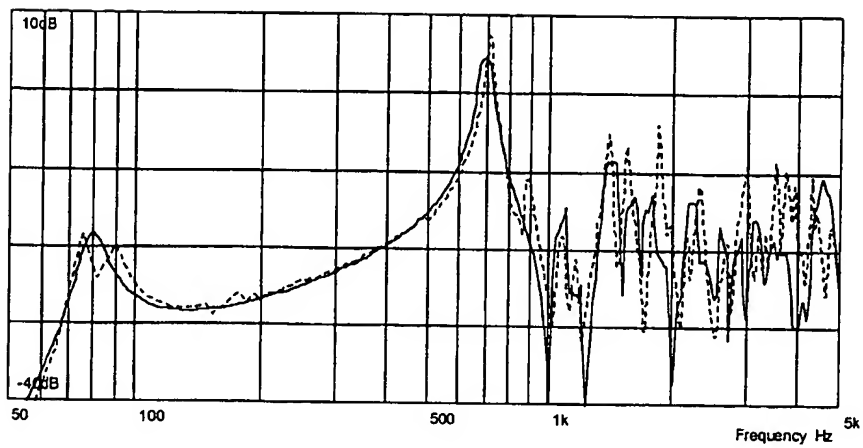
Fig. 25b
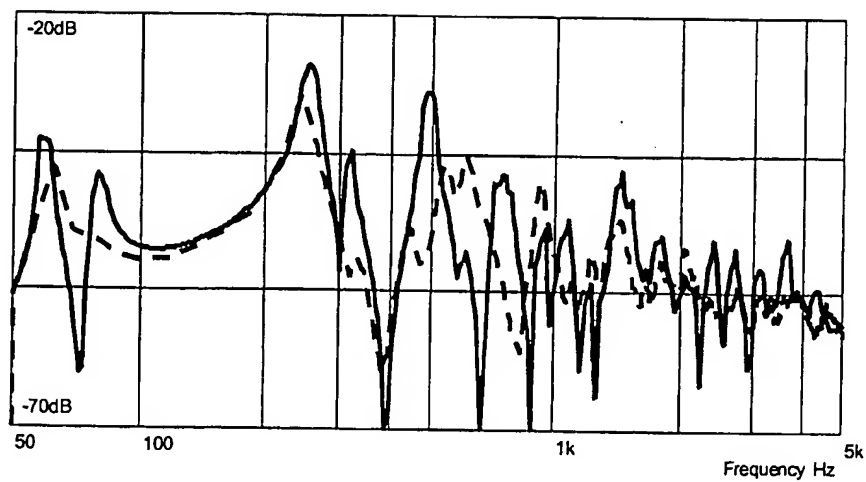
Fig. 26a free space
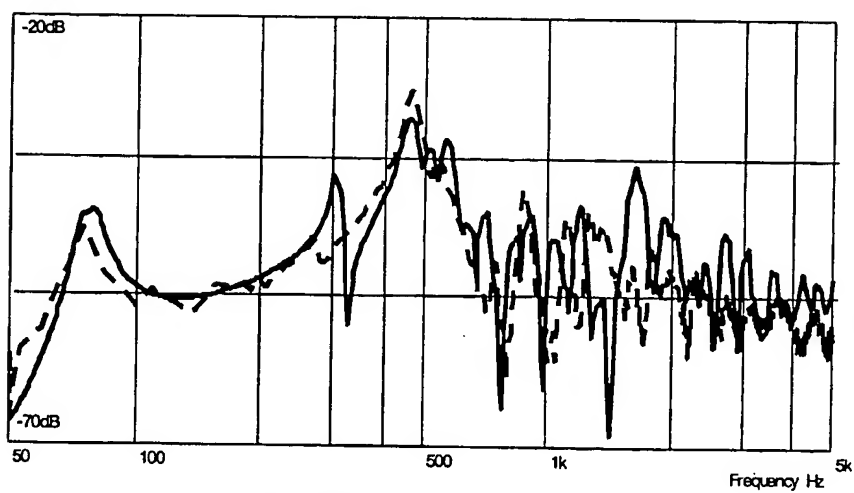
Fig. 26b Double enclosure
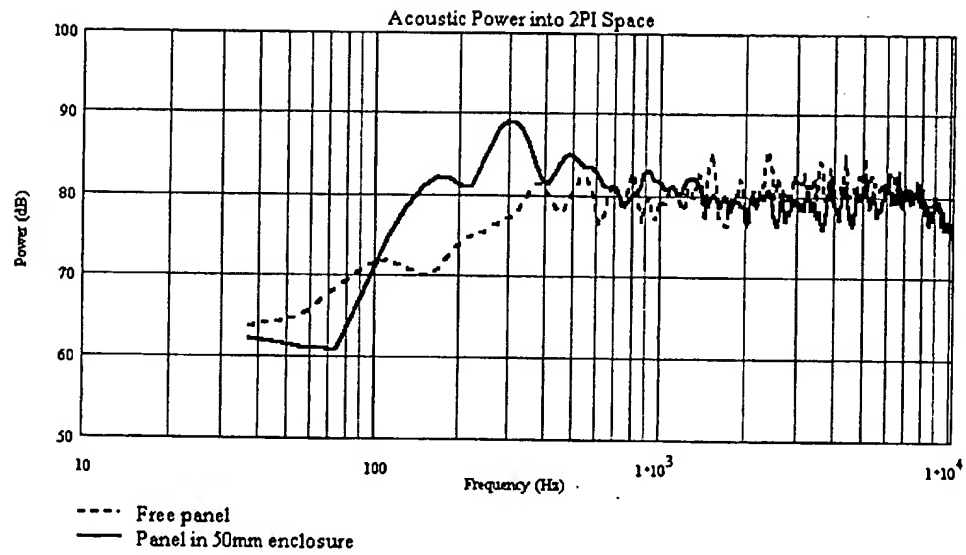
Figure 27a, A2-1 panel

Fig. 27b A2-1 panel

A2-1 Panel Power, 50mm Enclosure

Fig. 28a

A2-1 Panel Power, 95mm Enclosure
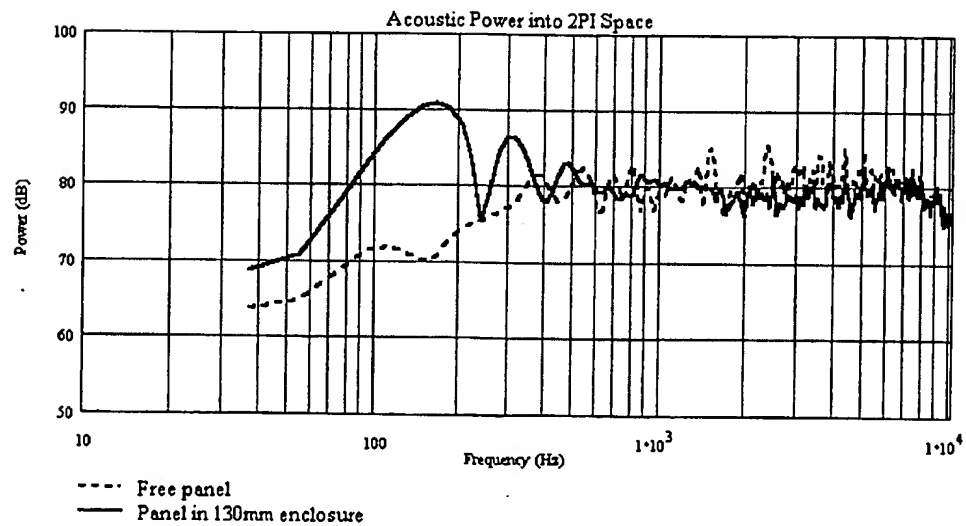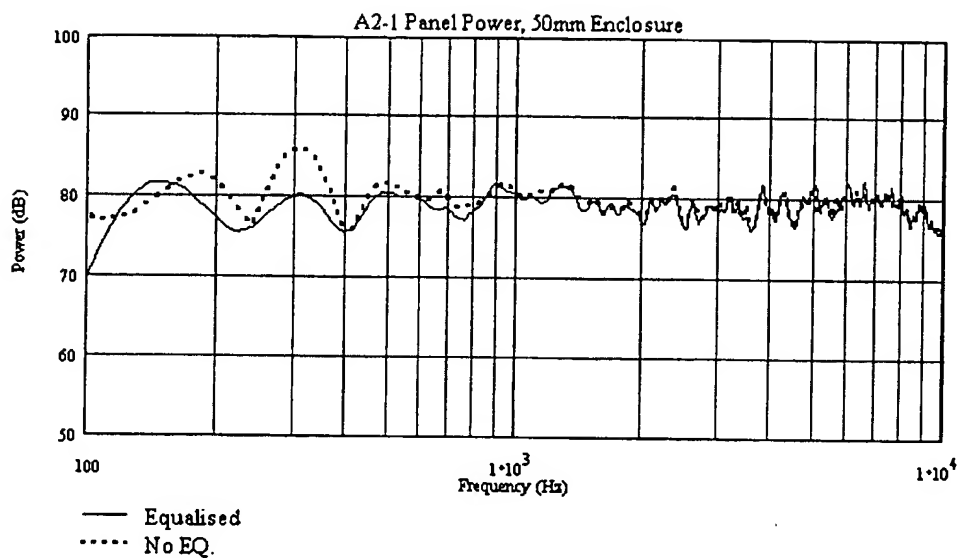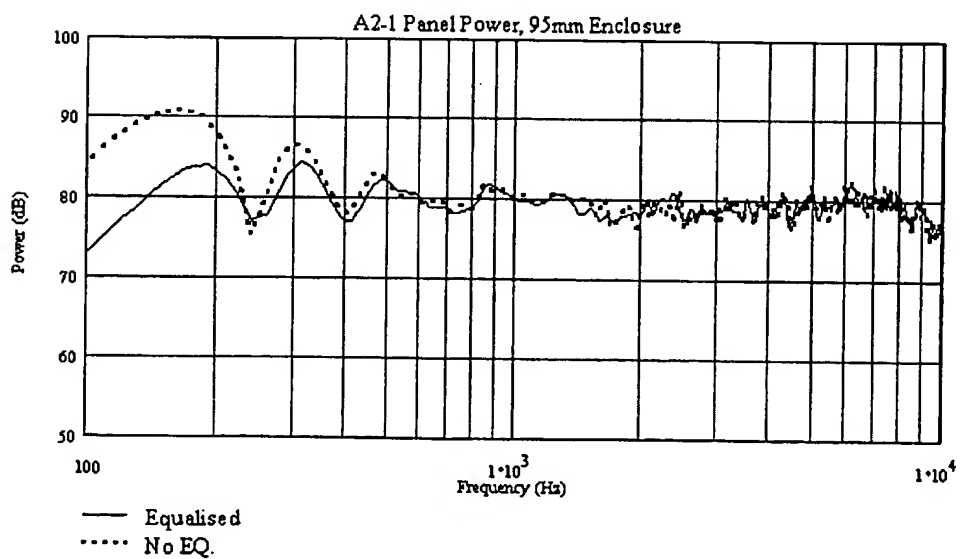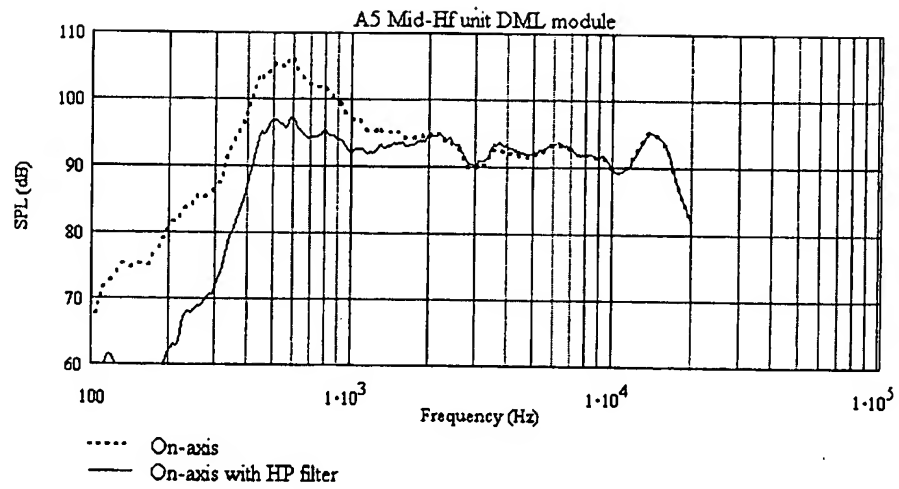
Fig. 28b

Fig. 28c.

INTERNATIONAL SEARCH REPORT

A. CLASSIFICATION OF SUBJECT MATTER
H 04 R 1/02

According to International Patent Classification (IPC) or to both national classification and IPC 6

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

H 04 R

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category°	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	WO 97/09861 A1 (VERITY GROUP) 13 March 1997, abstract, page 1, line 15 - page 4, line 5, fig. 3, claim 1. --	1,11
A	WO 89/00798 A1 (AVM HESS) 26 January 1989 (26.01.89), abstract, page 1, line 1 - page 4, line 17, fig. 32, claims 1,2. --	1,11
A	WO 97/09859 A1 (VERITY GROUP) 13 March 1997. ----	

☐ Further documents are listed in the continuation of box C.   ☐ Patent family members are listed in annex.

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Date of the actual completion of the international search	Date of mailing of the international search report
18 June 1999	09. 08. 1999

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax (+31-70) 340-3016	GRÖSSING e.h.

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ANHANG

zum internationalen Recherchen-
bericht über die internationale
Patentanmeldung Nr.

ANNEX

to the International Search
Report to the International Patent
Application No.

ANNEXE

au rapport de recherche inter-
national relatif à la demande de brevet
international n°

PCT/GB 99/01048 SAE 230542

In diesem Anhang sind die Mitglieder
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nannten internationalen Recherchenbericht
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Diese Angaben dienen nur zur Unter-
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This Annex lists the patent family
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cited in the above-mentioned inter-
national search report. The Office is
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Im Recherchenbericht angeführtes Patentdokument Patent document cited in search report Document de brevet cité dans le rapport de recherche	Datum der Veröffentlichung Publication date Date de publication	Mitglied(er) der Patentfamilie Patent family member(s) Membre(s) de la famille de brevets	Datum der Veröffentlichung Publication date Date de publication
WO A1 9709861	13-03-1997	AT E 176826	15-03-1999
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		AT E 177282	15-03-1999
		AT E 177574	15-03-1999
		AT E 177575	15-03-1999
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		AU A1 68815/96	27-03-1997
		AU A1 68816/96	27-03-1997
		AU A1 68818/96	27-03-1997
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